Chapter 1
Introduction

1-1. Purpose

This manual provides guidance in the structural, mechanical, and electrical design of lock gates and operating equipment at navigation projects.

1-2. Applicability

This manual applies to all HQUSACE elements, major subordinate commands, districts, laboratories, and field operating activities having responsibilities for the design and construction of civil works projects.

1-3. References

References are listed in Appendix A.

1-4. Applicable Computer Programs

CMITER, Computer Aided Structural Engineering, U.S. Army Engineer Waterways Experiment Station, 3909 Halls Ferry Road, Vicksburg, MS 39180-6199.

1-5. Plates

Illustrative plates containing general information, typical details, mechanical design data, and sample computations are included in Appendix B, and are referred to herein as Plate B-1, B-2, etc.

1-6. General

a. Function of gates. Lock gates serve a number of different functions, depending on location and conditions. While the major use of lock gates is to form the damming surface across the lock chamber, they may also serve as guard gates, for filling and emptying the lock chamber, for passing ice and debris, to unwater the lock chamber, and to provide access from one lock wall to the other by means of walkways or bridgeways installed on top of the gates. A navigation lock requires closure gates at both ends of the lock so that the water level in the lock chamber can be varied to coincide with the upper and lower approach channels. The sequence of "locking" a vessel upstream is: first, lower the water level in the lock to the downstream water level; second, open the lower gate and move the vessel into the lock chamber; third, close the lower gate and fill the lock chamber to the level of the upper pool; and finally, open the upstream gate and move the vessel out of the lock. Lockage of a vessel downstream involves a similar sequence in reverse order.

b. Types of gates covered.

(1) Miter gates. A very large percentage of the locks in the United States are equipped with double-leaf miter gates which are used for moderate- and high-lift locks. These gates are fairly simple in construction and operation and can be opened or closed more rapidly than any other type of gate. Maintenance costs generally are low. A disadvantage of this gate is that it cannot be used to close off flow in an emergency situation with an appreciable unbalanced head.

(a) Miter gates fit into recesses in the wall in the open position. The bottom of the recess should extend below the gate bottom to preclude operating difficulties from silt and debris collection. Enlarged recesses are sometimes used to facilitate the removal of accumulated ice. An air bubbler system is recommended to help clear ice and debris from gate recesses. (See Appendix D for typical air bubbler recess flusher.)

(b) Miter gates are framed either horizontally or vertically. The skin plate of a horizontally framed gate is supported by horizontal members which may be either straight girders acting as beams, or circular arches. Each such horizontal member is supported by the vertical quoin post at one end and the miter post at the other. All water load is transmitted through the girders and quoin blocking into the gate monoliths. A vertically framed gate resists the water pressure by a series of vertical girders more or less uniformly spaced throughout the length of the gate, and supported at top and bottom by horizontal girders transmitting the loads to miter and quoin at the top of the leaf, and directly to the sill at the bottom.

(c) The relative costs of the two types of gates (horizontally and vertically framed) depend largely upon three main factors:

- Overall weight of the gate;

- Simplicity of design and ease of fabrication and erection;

- Cost of that part of the lock walls and sills influenced by the design of the gates.

(d) When the ratio of the height of a leaf to its width is greater than about 0.7, the horizontally framed gate will weigh less. For long, shallow gates, vertical framing requires less material.

(e) The overturning moment carried to the lock wall by a horizontally framed gate is greater about all points below the sill than that caused by a vertically framed gate, unless the entire sill load is transmitted to the wall. Hence, the latter type requires less masonry in a thrust wall of gravity section, but the heavier sill necessary to support the bottom girder into which the verticals are framed may counterbalance this saving.

(f) Due to the greater rigidity and resistance to boat impact of the horizontally framed gate and the insignificant difference in cost, the vertically framed gate will no longer be used except for unusual applications and upon special approval.

(2) Sector gates. A sector gate is similar in shape to a tainter gate except it is oriented to rotate about a vertical axis and is supported at the top and bottom in a manner similar to a miter gate. Like miter gates, sector gates are used in pairs, meeting at the center of the lock when in the closed position and swinging into recesses in the lock walls for the open position. The trunnions are located in the lock walls, and the skin plates face in the direction of the normally higher pool level.

(a) Sector gates are used at both ends of locks that are located in tidal reaches of rivers or canals where the lifts are low and where the gates may be subjected to reversal of heads. Since these gates can be opened and closed under head, they can be used to close off flow in an emergency. The gates swing apart and water flows into or out of the lock through the center opening between the gates. In some cases, flow is admitted through culverts to improve filling characteristics or where ice or drift may not permit adequate flow between the gates.

(b) Because the turbulence area at the upper end of a lock filled by a sector gate is not effective for lockage of vessels, the length of the lock chambers must be increased proportionately. Model tests indicate that about 100 feet (ft) of additional length is required. Like other end-filling systems, sector gates cannot be used for filling and emptying high-lift locks unless the filling and emptying rates are greatly reduced. The practical lift limitation is usually about 10 ft, although gates with higher lifts have been built.

(c) The disadvantages of the sector gates are high construction cost, long opening and closing times, and larger wall recesses.

(3) Vertical-lift gates. Vertical-lift gates may be used at both ends of a lock, or at only one end in combination with a miter gate at the other end. They can be raised or lowered under low to moderate heads but are not used when there is reversed head. Their operation time is much slower for older gates and maintenance costs are higher than those of miter gates, but they can be used in emergency closure. The newer gates, however, are capable of achieving operating speeds equal to, or even faster than, miter gates.

(a) A vertical-lift gate installation at the upstream end of a lock normally consists of a single-leaf submergible gate, which rises vertically to close off the lock chamber from the upper pool. When the lock is filled, the gate is opened by sliding the leaf vertically downward until the top of the leaf is at or below the top of the upper sill.

(b) In some cases, a double-leaf vertical-lift gate may be used. The upper leaf can be provided with a curved crest which permits overflow to supplement flow from the primary filling system when the lock chamber is nearly full. This type of gate can also be used for skimming ice and debris.

(c) When a vertical-lift gate is used at the downstream end of a lock, it is raised vertically to a height above the lower pool level so that vessels can pass underneath. The gate leaf is suspended from towers on the lock walls and may be equipped with counterweights to reduce the power hoist size. Lock gates of this type are practical only for very high locks and where required vertical clearance can be provided under the gate in its raised position.

(4) Submergible tainter gates. The locks of the Dalles Dam, some Lower Snake River projects, and the Upper and Lower St. Anthony Falls Locks have submergible tainter gates. This type of gate is raised to close the lock chamber and lowered into the lock chamber to open it. The end frames are recessed into the lock wall so no part of the end frame projects into the passageway. This type of gate was chosen because it is structurally efficient and was estimated to be lighter in weight and less costly than a double-leaf miter gate for these applications. Also, the tainter gate permitted the length of the approach channel to be reduced by the leaf

width of the miter gate. There are two potential problem areas in the operation of this type of gate: skewing of gate during opening and closing, and vulnerability to damage if hit by lock traffic. However, with good design practices and lock management, these problems will be minimal.

1-7. Materials and Working Stresses

a. Materials. This manual serves only as a guide and the following list should not be considered as a complete listing of materials that may be used.

(1) Structural steel. Lock gates are usually constructed of structural grade carbon steel having a yield point of 36,000 pounds per square inch (psi). Low-alloy steel, with a yield point up to 50,000 psi, is quite frequently used as skin plate in conjunction with structural grade girders. In some cases for the larger gates, other than miter gates, low-alloy steel may be economical for the complete gate. The deflection of members fabricated of high-strength low-alloy steel should always be investigated as it will always be more severe than if the members were of structural grade carbon steel. For miter gates, structural grade carbon steel should be used regardless of the gate height in the interest of providing a more rigid gate, except for the skin plate and diagonals which may be of high-strength, low-alloy steel whenever warranted.

(2) Corrosion-resistant steel. Corrosion-resistant steel normally should only be used in locations where corrosion is expected to be severe or where corrosion will impair the normal efficiency of gate operation. Under most conditions, seal contact surfaces of lock gates are not required to be corrosion resistant but under adverse conditions corrosion-resistant clad or solid stainless steel plates may be desirable. For some lock locations and conditions it may also be desirable to clad the contact surfaces of miter and quoin blocks with corrosion-resistant steel or to use solid stainless or corrosion-resistant steel miter and quoin contact blocks. Flame spraying of corrosion-resisting steel particles to surfaces subject to severe corrosion may be advantageous where using solid or clad corrosion-resisting steel is not practical or not economical. The new guide specification CW-05036 covers the requirements for surface preparation and applications of metalizing/flame spraying coatings.

(3) Cast steel. The operating strut pin bearing collars, pintle sockets, and pintle shoes are normally fabricated of cast steel, utilizing mild-strength to medium-strength carbon steel castings. For items that are subjected to higher stresses than medium-strength castings are capable of carrying, such as the miter guide roller and pintle balls, high-strength, low-alloy steel castings should be used.

(4) Forged steels. Gudgeon pins, operating strut connecting pins, anchor link pins, parts of the anchorage links, and guide roller pins should be made of carbon steel forgings rated for general industrial use. Forgings may be untreated or heat-treated depending on intended use and requirements. The pintle ball of most gates is made of an alloy steel forging containing nickel, giving the forging a good allowable bearing value as well as a fair degree of corrosion resistance. Corrosion-resistant weld overlays may also be used on pintle balls in highly corrosive environments.

(5) Bronze. Bushings for all lock gate components are normally made of bronze. Usually bearings are made of a bronze designated for general purpose applications. Where stresses are encountered that are higher than desirable for the general purpose bronze, aluminum bronze may be used.

(6) Bolts. Where bolted connections are used for parts of the gate that may have to be removed for maintenance or repair, a copper-nickel alloy, usually referred to as monel, or an equally corrosion-resistant steel should be considered, especially if corrosive elements are present. The 300 series stainless steel and bronze bolts, nuts, washers, and setscrews have been used with good results in highly corrosive environments. Ordinarily, bolts, nuts, and washers should all be made from the same type of material; however, if salvage and reuse is intended different alloy combinations for bolts and nuts should be used to minimize seizing. Normal applications of this type of connection are pintle socket to gate connection, quoin and miter water seal bolts, and bolts for the bottom seal. Where bolts are used and corrosion is not a factor, ASTM A307 or A325 bolts should be used, with bolt strength dictated by load and conditions.

(7) Fabrication. Fabrication of all lock gates should be by welding, with bolts being used only for those parts that may have to be removed for maintenance or repair. The application of welding generally results in lighter and stronger gates. All welding should be done in accordance with the current Structural Welding Code of the American Welding Society, Section 9, Design of New Bridges.

b. Design strength. Structural steel gate members shall be designed in accordance with the requirements of EM 1110-2-2105. For general purpose bronze bushings, the allowable bearing should be below 1,500 psi with a maximum concentrated bearing of not more than 5,000 psi (this refers to a bushing with an eccentric load). Where a higher allowable bearing is desirable, aluminum bronze may be used with working stresses up to 5,000 psi and a concentrated bearing of not more than 10,000 psi as described above. Working stresses for both forgings and castings should be based on yield strength and for normal applications should be no more than $0.50F_y$.

1-8. Basic Dimensions

a. Miter gates. A miter gate is a three-hinged arch when the leaves are mitered. Gate geometry is a function of the angle the work line of the leaf makes with a line normal to the lock walls, with the gate in a mitered position. Past study and design have determined that for miter gates a slope of 1L on 3T gives the best results (L = longitudinal, T = transverse). In general, vertically framed gates have been used where the height-to-length ratio of the leaf was less than 0.5. The approximate ratio of height to length, where the weight of a vertically framed leaf is essentially the same as a horizontally framed leaf, is somewhere between 0.70 and 1.0 (see paragraph 1-6*b*(1)). However, vertically framed gates are not recommended for new construction. Even with a slight increase in cost, the greater rigidity of the horizontally framed gates makes them more desirable.

(1) The pintle is located so that the leaf, when recessed, is completely within the lock wall and so that the pintle is eccentric (upstream) with respect to the center of curvature of the bearing face of the quoin contact block. The center of curvature of the bearing face is always located on the line tangent to the thrust line at the quoin contact point. The pintle eccentricity, which makes the quoin block approach the contact point tangentially, should be approximately 7 in., thereby reducing the possibility of metal interference (see paragraph 2-1*h*).

(2) The arch-type miter gate is a horizontally framed structural system of curved members with a composite acting skin plate. Except for the curvature, the gate size and other components are similar to or the same as horizontally framed straight gates.

b. Sector gates. Sector gates are generally laid out with the frames forming an equilateral triangle. The normal layout is for 60 degrees (deg) or greater interior angles, formed by the frames and a chord line behind the skin plates. One strut is parallel to the lock wall in the closed position, thereby causing the other strut to form an angle equal to the interior angle, with the lock wall. The pintle is located so that the gate is completely in the recess in the open position.

c. Vertical-lift gates. Dimensions for vertical-lift gates are based solely on lock width and girder depths required by the head. Recesses in lock walls for upper gates are determined by load, girder depth, and detail requirements. Towers for downstream gates are also determined by load, counterweights, and related details.

d. Submergible tainter gates. As with spillway tainter gates, the controlling dimensions are the lock width, gate radius, and end frame and trunnion hub location. Plates B-45 and B-46 show typical end frame and cross section of the gate.

1-9. Loads

The loads applicable to lock gate design are dead, hydrostatic, hydraulic, temporal, and boat impact. Miter gates are also subject to torsion. Dead load is the weight of the structure plus mud and ice; hydrostatic load is the water load on the gate produced by the pool differential; boat impact is the dynamic force applied to the gate by the barge impact; temporal load is the water surge forces from wave loads or overfilling of the lock; and torsion on miter gates is the result of a twisting action from the operating strut force and the water resistance caused by the leaf moving. For more specific lock gate loads, see the gate type's respective section herein.

a. The controlled upper and lower pools cause the normal loading, and greater water forces, such as unwatering the lock, are considered emergency conditions, with an increase in allowable stress of 33 percent.

b. The force of impact usually is limited by local failure in the region of impact. For design purposes, this force, supported by past designs, is converted into an equivalent water load of 10- to 15-ft head below the top girder and of 6- to 10-ft head above the top girder for vertical-lift gates if horizontally framed. An impact load of 250,000 or 400,000 pounds (lb), according to location of load, is applied above the pool to horizontally framed miter gates. (See paragraph 2-1*b*(1)(d).) Barge or impact force is generally not applied to vertical framing members. Greater impact forces or the use of barriers may be justified based on the importance of the waterway or type of traffic.

c. On sector gates a design withstanding a concentrated impact force of 125,000 lb applied to the top horizontal girder is recommended, with vertical framing members designed for no impact loading.

d. The quoin end of a miter gate leaf is held vertical by the pintle and gudgeon pin, leaving the miter end free to twist out of vertical alignment. The deadweight of the leaf, along with ice or mud, also causes the leaf to twist. To keep the leaf in vertical alignment while stationary, and to eliminate excessive deflection during operation, diagonals are provided on the downstream faces of horizontally framed miter gates and on both upstream and downstream faces of vertically framed miter gates. These diagonals act as tension members for all normal gate operations. (See USAED, Chicago 1960.)

1-10. Fatigue and Fracture Control

All possible modes of failure should be considered when designing lock gates. Possible failure modes are: 1) general yielding or excessive plastic deformation, 2) buckling or general instability, 3) subcritical crack growth leading to loss of cross section or unstable crack growth, and 4) unstable crack extension leading to failure of a member. Failure modes 1 and 2 are addressed by Load and Resistance Factor Design (LRFD) and Allowable Stress Design (ASD) principles whereas failure modes 3 (fatigue) and 4 (brittle fracture) can be addressed using fatigue and fracture mechanics principles. Welded construction with its emphasis on monolithic structural members has led to the increased desirability of including fracture criterion in addition to strength and buckling criteria when designing a structure. Stress range, detailing, and the number and frequency of load cycles control fatigue while geometry, toughness, and stress levels control fracture. For further guidance, see EM 1110-2-2105.

a. Fatigue requirements. Fatigue can be controlled by stress range, detailing, and the number and frequency of load cycles. While the number and frequency of load cycles are usually controlled by the structure's purpose, the designer can control the stress range and the choice of detail. Refer to AISC (Current Edition), Appendix K, for guidance in design and detailing.

b. Fracture control requirements. The designer should set limits on tensile stress levels, enforce controls on quality fabrication and inspection procedures to minimize initial defects and residual stresses, designate the appropriate temperature zone, and specify the related minimum fracture toughness for critical members and/or components. For lock gates, fracture critical members shall be defined as "members and their associated connections subjected to tensile stresses and whose failure would cause the structure to be inoperable." For minimum Charpy V-notch impact test values see EM 1110-2-2105.

1-11. Operating Machinery General Design Criteria

a. Machinery components. All components of the gate operating or hoisting machinery except compression members which may fail by buckling should be designed for loads or forces produced by an effective cylinder operating pressure or normal full load torque of an electric motor with a minimum safety factor of five based on the ultimate tensile strength of the material involved. In addition, each part or component should be designed for a unit stress not to exceed 75 percent of the yield strength of the material for the maximum load, maximum cylinder pressure obtainable, or overload torque from an electric motor.

b. Piston rods. Piston rods and other compression members in which failure may be caused by buckling should be designed in accordance with either the Johnson or Euler equation, whichever applies. A factor of safety of at least 2.5 should be provided based on the maximum load to be imposed on the member and the critical buckling load. In almost all cases the end fixity coefficient for pin-ended columns should be used.

c. Shafting. Shafting should be designed for the rated loads, increased by applicable shock and fatigue factors, with a factor of safety of five based upon the ultimate strength of the materials, provided the stresses produced by the maximum torque of the motor do not exceed 75 percent of the yield point of the materials involved. Stress concentration factors should be used where applicable. A combined shock and fatigue factor of 1.25 should be used. Shafting should be amply supported, and provided with adequate means to prevent longitudinal movement. The distance between bearings on shafting subject to bending, except that due to its own weight, should be such that the maximum shaft deflection will not exceed 0.01 in./ft of length at rated load. Torsional shaft deflection should not exceed 0.08 deg/ft of shaft length at rated load. If spur gears are mounted on the shafts it is necessary to limit the relative slope of the shafts containing the gear and pinion. It has generally been found acceptable to limit the slope of the shaft at the center line of the gear mesh to one-third the backlash divided by the gear face width. The usual range of backlash for spur gears is 0.03/D.P. to 0.05/D.P. in., where D.P. is the diametral pitch.

d. Speed reducers. Speed reducers should be worm, helical, or herringbone type in accordance with the applicable American Gear Manufacturers Association (AGMA, Current Edition) standards with antifriction bearings. If possible, an oil meeting the requirements for the ambient temperatures that will be encountered should be used. Where ambient temperature range will exceed that recommended for the oil, a thermostatically controlled heater should be provided in the reducer case to keep the oil at the temperature recommended by the oil manufacturer. Where heaters are used, the surface area of the heater should be as large as possible to prevent charring of the oil. The watt density of elements selected should not exceed 10 watts per square inch. In the interest of energy conservation, consideration should be given to insulating the reducer case to minimize heat loss. Another alternative would be the use of a synthetic gear lubricant with a minus 40° Fahrenheit (F) pour point if acceptable to the reducer manufacturer. Reducer selection should be based on manufacturer's published ratings for the required service conditions.

e. Couplings. Flexible couplings should be of the gear type. Couplings should have flanged sleeve housings and integral lips at each end to house the seals and retain the sleeves. Selection normally should be based on manufacturer's published rating. Sleeves should be fastened so that they cannot work loose or slip off. Couplings with sleeves held in place with snap rings should not be permitted.

f. Brakes. Brakes should be of the shoe type, spring set, with D-C magnet operated release and should be completely enclosed in a watertight and dusttight enclosure. The brake should have a torque rating not less than 150 percent of the full load torque of the motor when referred to the shaft on which the brake wheel is mounted, efficiency not being considered. The torque rating should be based on continuous duty. Fuses should not be used in the brake control circuit.

g. Bearings.

(1) Antifriction bearings should be selected in accordance with manufacturer's published catalog ratings. Life expectancy should be based on 10,000 hours B-10 life with loads assumed equal to 75 percent of maximum.

(2) Bronze sleeve bearings should have allowable unit bearing pressures not exceeding the following:

(a) Sheave bushings, slow speed, Federal Specification QQ-C-390B, Alloy C90500, 3,500 psi.

(b) Main pinion shaft bearings and other slow-moving shafts, hardened steel on bronze Federal Specification QQ-C-390B, Alloy C90500, 1,000 psi.

(c) Bearings moving at ordinary speeds, steel or bronze Federal Specification QQ-C-390B, Alloy C93400, 750 psi.

h. Open gearing. Open gearing should have spur teeth of the involute form, to comply with AGMA 201.02, ANSI Standard System, "Tooth Proportions for Coarse-Pitch Involute Spur Gears" (Information Sheet A). Strength should be based on static load from the Lewis equation modified for pitch line velocity by the factor (600 + velocity in feet per minute (fpm)) divided by 600.

i. Efficiency. In computing losses in a lock-gate-operating machine the following should be used as a guide:

(1) Silent chain (includes oil-retaining and dust-tight case)	97%
(2) V-belt (includes both drive and driven sheave)	90-96%
(3) Spur gear reduction unit up to	
16:1 ratio	88%
16:1 to 40:1 ratio	84%
14:1 to 150:1 ratio	78%
(4) Herringbone gear reduction unit	
Single reduction	97%
Double reduction	95%
Triple reduction	90%
(5) Planetary or helical reduction unit	
Single reduction	97%
Double reduction	95%
Triple reduction	90%
(6) Pair of spur gears (Gears only)	97%
(7) Pair of bevel gears (Gears only)	95%
(8) Worm gear reduction unit	

Since worm gears are the most controversial class of all gearing, the manufacturers should furnish the certified starting and running efficiency of the unit, particularly if the unit is operated at other than standard speeds.

(9) Bearings

Ball and roller	98%
Intermediate horizontal shafts, bronze bushings	95%
Very slow speed shafts, bronze bushings	93%

j. Hydraulic systems and components.

(1) System types. Two basic types of lock hydraulic systems are currently in use. One is the central pumping unit type where the system pumps are all located in one central location with supply and return extending to the gate and valve operating machinery location on the lock. The other type consists of pumping unit assemblies complete with reservoir, valves, and necessary system components located at each individual operating cylinder, or at each of the four corners of the lock. A pumping unit at each corner could then operate a gate leaf and a tainter valve. If local pump systems are used at each operating cylinder, they could also be piped to a near valve or gate cylinder for backup hydraulic power.

(a) Central pumping system. Many of the central pumping systems of the past have used constant displacement screw pumps with system operating pressures of 900 psi to as high as 1,500 psi. Usually, three main supply pumps and one smaller capacity holding pump were used. The capacity of the main pumps was such that for normal operation two pumps would supply the required flow. Operation was alternated between the three pumps so as to equalize wear, while also maintaining standby capability. The smaller capacity pressure holding pump was used to build and maintain system pressure and allow the larger pumps to start unloaded and operate unloaded when the gates were not being moved. With this system, flow control and deceleration valves were used to control the speed of the valves or gates. Experience has shown this system to be simple, reliable, and fairly economical. In recent years variable displacement piston pumps have been used with the central pumping system. Pumps with three to five preset delivery positions have been used, with one position set at zero delivery so that the pumps can start or idle unloaded. Systems using these pumps rely on the preset variable displacement to control valve and gate speeds. With these types of pumps, two pumps are required for each lock so that each miter gate leaf cylinder can be supplied by one pump. With a double lock, the four required pumps provide adequate standby capacity through interconnecting valving. On a single lock a third standby pump should be considered; however, if economics or space requirements preclude installation of the third pump, the two pumps should be interconnected to provide standby operational capability at reduced speed, if one unit should malfunction. The variable displacement capability makes these pumps very well suited for controlling gate speed.

(b) Local pumping system. The local pumping system is usually used on locks that are not subject to flooding (overtopping); however, local pumping systems can be used with success on locks subject to inundation with special attention paid to lock design. Local hydraulic pumping units and controls should not be located in the galleries on locks which are subject to overtopping. Where galleries are used the galleries should be sealed with watertight doors; the piping should penetrate the walls through sealed sleeves; and a sump pump should be provided to handle any leakage incurred. Local pumping systems can utilize any of the conventional types of pumps, with operating pressures in the 2,000- to 3,000-psi range acceptable. Large system pressure drops and high system shock pressures are reduced by the absence of long hydraulic line runs. Variable displacement piston pumps may be best suited for this application due to their high pressure capabilities, efficiency, and variable flow capabilities.

(2) System operating pressure. Many factors must be considered in the selection of the system operating pressure. Among the most predominate of these factors are reliability, serviceability, efficiency, safety, and economics. Also among these factors are pump type, pressure rating and capacity, cylinder size and pressure rating, pipe size, friction loss, bursting pressure, and system shock. In Europe, operating pressures as high 5,000 psi have been used for several years on locks and dams with good success, whereas in the United States these pressures are still not as common with hydraulic equipment manufacturers. System operating pressures should be as follows:

(a) Central pumping system operating pressure of 900 psi to as high as 3,000 psi should be satisfactory.

(b) Local pumping system operating pressure of 1,500 psi to 3,000 psi should be satisfactory and desirable.

(3) System components. The manufacturer's published pressure rating should be used for the selection of all system components. All published ratings should be equal to twice the system's operating design pressure, thus establishing a high level of quality for the equipment.

(a) Cylinders. The types of hydraulic cylinders that have been used on locks in the past include the specially designed, the tie-rod type, and the mill type. Manufacturers' standard mill type is the preferred, as it is known as the extra heavy duty type. These, however, should be designed with a factor of safety of five based on the ultimate strength of the material involved. Tie rod cylinders have also been used with good success. Where specially designed cylinders are required, they may be constructed of seamless steel tubes on flat plates rolled into cylinders; forged in one piece with integral flanges; or centrifugally cast steel. Fittings for supply and return lines to hydraulic cylinders should be mounted on the top or sides of the cylinders. These connections should be for "SAE four-bolt flange" or "SAE straight thread O-ring" connections for installation and maintenance convenience. Cylinders should be fitted with air bleed vents and drains at each end of the cylinders. Another optional feature which may be beneficial under certain circumstances is the adjustable cushion which is used for deceleration control at stroke limits. Piston rods are usually chrome plated for wear resistance and may be high strength stainless steel where corrosion is a concern. Ceramic coated rods, which have been used for several years in Europe but are relatively new in the U.S., may be considered for corrosive and abrasive service. On cylinders that are foot mounted, wedges should be provided between cylinder feet and shear plates to assure a tight fit. Specifications should indicate that cylinders shall be shipped with piston rod retracted and restrained from movement. Cylinders should be filled with new hydraulic fluid (the same type as that specified for the system) after manufacture to prevent corrosion during the storage period prior to use. An accumulator charged to 100 psi connected to the rod end port to allow for fluid expansion and contraction is a good method to do this.

(b) Pumps. Several different types of pumps have been used in locks in the past; these include gear, vane, and piston (both axial and radial) pumps. The gear pump is simple in design, rugged, has a large capacity for a small size, is low in cost, and has a high tolerance for contaminants in hydraulic fluids. The gear pumps' low volumetric efficiency, high wear characteristics, noise, fixed volume, and relatively short life expectancy make them undesirable for main pressure pumps. Variable volume vane pumps are efficient and durable if a clean hydraulic system is maintained. They are generally quiet, but may whine at higher speeds, and they are compact in relation to their output. The piston-type pump is the one recommended for main hydraulic power, as it has the highest volumetric and overall efficiencies, is capable of high output pressures, readily lends itself to variable displacement, and generally has long life expectancy. In order to reduce noise and increase life expectancy the pump speed should be 900 to 1,800 rpm. A variable delivery radial piston type, three to five adjustable delivery rate pump with solenoid control for selection of pumping rates is a desirable pump for main hydraulic pressure supply. One of the delivery rates on the pump should be set at neutral or zero delivery so as to start the pump motor under a no-load condition. The individual controls on each pump should be adjustable from zero to full flow capacity at each control setting so that flow rates can be varied in the field to suit minor variations in operating conditions. The pump shall be equipped with an auxiliary gear pump for pilot pressure, internal pressure relief valves, and an adjustable flow control device to control the speed of shifting between pumping rates. Variable volume axial piston pumps with the swiveling barrel, rather than swashplate, have also been used with good success. These tend to be a higher grade pump with less noise, vibration, and wear than the swashplate design. If an axial piston pump is used, then an additional auxiliary pump will be required for pilot pressure.

(c) Directional control valves. The directional valves normally used are four-way, three-position, blocked center solenoid-controlled pilot operated, spring-centered type. Where the solenoid control is selected, the valve should be equipped with adjustable orifices to slow the action of the spool when changing spool positions. The edges of the valve spools should be grooved to provide a throttling effect when moving from one position to another. The tandem center-type spool has been used in the past; however, system diversity is limited by this type of valve. Lever-operated directional control valves have also been used in the past; but they will not lend themselves to interlock logic control, so they should be considered on only the most basic application. The directional control valve could be the single greatest pressure loss point in a hydraulic circuit and, therefore, should be given a great deal of design attention. If practical, the directional control valves should be designed for 1.5 to 2.0 times the maximum flow rate required in order to minimize pressure loss. There are several companies that manufacture high quality manifolds for mounting cartridge-type control valves. This type of system can be used to mount directional control, relief valves, counterbalance valves, proportional valves, and other types of control valves. The manifold system is economical, minimizes pipe fabrication, possibility of leaks, and space requirements and, therefore, should be used wherever possible. The solenoid-control valve for the main directional control valve should be provided with manual operating pins for emergency operation during solenoid malfunction or failure.

(d) Relief valves. Relief valves that are normally used on pressure lines are the balanced piston type, internally operated with an adjustable operating range. These valves have been used with good success and have proven to be rugged and reliable. The manifold and cartridge valve system should be considered as it may prove to be advantageous because it offers space and cost savings and easy maintenance. The cartridge valves should be the pilot-operated poppet type with adjustable pressure relief range. Response times are very quick on cartridge valves, usually in the 10-millisecond range, so the designer must decide how much slower the poppet should respond to eliminate the operational shocks.

(e) Flow control valves. Flow control valves, if required, should be of the adjustable orifice type, allowing controlled flow in one direction, free flow in the other direction. Where inertia loads such as miter gates are being controlled by hydraulic cylinders, the flow control valve should be placed to control the oil leaving the cylinder. When used in conjunction with a counterbalance valve, or in an element moving vertically, the flow control valve should control the oil entering the cylinder. Here again the manifold and cartridge valve system should be considered.

(f) Reservoirs. Hydraulic fluid reservoirs should have a minimum capacity in gallons of about three times the maximum pump capacity in gallons per minute (gpm). There are other factors also that must be considered in sizing of a reservoir. If the reservoir is cross-connected to another system to serve as an emergency backup, then an analysis of the potential “overfill” or “overempty” of the reservoir must be accomplished. This is due to the increase or decrease in the volume of the operating cylinder to which it is cross-connected. Long line runs and thermal expansion of the fluid must also be taken into consideration when sizing the reservoir. In any case, the reservoir should have a capacity to always provide a flooded suction to the pump. The interior of the reservoir should be coated with a good epoxy coating system suited for hydraulic service. The reservoir top, sides, and bottom should be fabricated of heavy steel plate, 1/4 in. to 3/8 in. thick, annealed and pickled. Internal reinforcing should be provided to ensure sturdy mounting for the pump unit, with vertical oil baffles to separate oil return from the pump inlet, and to provide a nonturbulent flow of oil to the pump suction. Consideration should be given to providing vibration isolation between the pump base and the oil reservoir to minimize noise transmission. Reservoir accessories such as suction filter, oil level gage, low level shutoff switch, magnetic particle unit, drain valve, removable clean-out plates (both ends), reservoir heaters (if required), and replaceable filter breather cap should be provided. Where reservoir heaters are used, the watt density should not exceed 10 watts per square inch. When a free-standing reservoir is used with a central system, it should be of such size as to promote cooling and contaminant separation and allow thermal expansion of the fluid and changes of fluid level due to system operation. The design minimum fluid level should be high enough to prevent vortex formation at the pump inlet opening. Adequate pump suction submergence should be available from the pump manufacturer. Many central systems have been built with free-standing tanks of approximately 1,000-gallon (gal) capacity.

(g) Filters. To provide initial cleanup and continuous filtering of the hydraulic oil, a full flow, removable-cartridge-type oil filter should be provided in the return line. Pressure gages should be installed on the filter tank to indicate pressure drop across the filter. Also, a filter cartridge replacement indicator would be beneficial to maintenance personnel. A system should be provided to indicate when oil is bypassing the filter. Filter elements should be capable of removing particulate matter of 10 micrometers, with a filtration ratio (Beta) of $B_{10} = 75$. The Beta (B_n) is the ratio of the number of particles greater than a given size (n) in the influent fluid to the number of particles greater than the same size (n) in the effluent fluid. ANSI B93.30M should be referenced for the B_{10} filtration ratio test procedure. Since proper operation of the control valves, relief valves, and pumps depends on the cleanliness of the oil used, it is a very important consideration in the design of the hydraulic system. Very rigorous specifications should be prepared requiring recirculation of all oil in the system through the system filter before the unit is put into operation. When a separate pilot pressure pumping system is provided, an in-line, system pressure filter with a replaceable cartridge should be furnished in the pump discharge line.

(h) Accumulators. Accumulators should be used in systems with long lines to minimize the effect of system shock pressures.

(i) Piping. Piping for hydraulic systems which are free from appreciable shock, vibration, and external load should be designed for a safety factor of six based on the operating pressure. Where severe shock, mechanical abuse, or vibration are likely to occur, then a safety factor of eight should be used. Where flexible hoses are used they should be designed with a minimum safety factor of eight. Also, exposure to the elements, any abnormal equipment hazard, and chafing are elements to be considered in the design of flexible hoses. Normally, piping should be seamless, black steel pipe, pickled and oiled with forged steel weld fittings. Piping 2 in. and smaller should use socket-weld-type fittings, and over 2-in. piping should use butt-weld type. Where piping crosses under the lock chamber, or other applications subject to very corrosive conditions, stainless steel pipe and fittings should be used. Hydraulic tubing with flare- or swage-type fittings may be used with package-type pump units. Expansion joints are normally not required, or recommended; however, every piping system should be analyzed and adequate provisions made for pipe expansion and movement. Pipe hangers should be of a type that are not rigidly connected, in order to prevent breakage from line shock or pipe movement within the hanger. Some desirable features that should be considered for maintainability of a hydraulic piping system are:

(1) Piping should be pitched a minimum of 1/2 in. per 50 ft in order to provide high and low points.

(2) Air bleed valves should be provided at high points in the system.

(3) Drain valves should be provided at low points in the system.

(4) Periodic shutoff valves should be provided, especially in long runs, to facilitate maintenance without complete system drainage.

(5) Periodic pressure gage ports with gage cocks should be considered for installation of gages for system troubleshooting.

With the central pumping system, supply and return lines at the ends of long runs of piping should have a valved cross-connection to permit start-up, or periodic flushing. Piping for hydraulic systems should be hydrostatically tested at 150 percent of system design pressure.

(j) Fluid velocity and pressure drops. The selection of fluid velocity for the computation of pipe friction is, in general, a compromise between limitations of pressure drop and limitations of line size. The following paragraphs indicate the general range of velocities acceptable.

(k) Pressure supply lines. The velocity in supply lines is generally held between 10 and 15 feet per second (fps), although for very short lines, as used in the package-type unit, velocities up to 20 fps are not considered excessive.

(l) Return lines. Generally the velocity in return lines is kept basically the same or slightly less than the velocity in supply lines.

(m) Pump suction. Velocities in pump suction lines normally are in the range of 2 to 5 fps. Pump suction lines deserve special attention to velocity since the pressure drop in the line together with the pressure drop through a suction filter, if used, can be detrimental to pump performance, especially during cold start-up. Excessive pressure drop in the pump suction line is a frequent cause of pump cavitation.

(n) Drain lines. Although flow in the drain lines from the valves is normally very small, velocity must be kept low to avoid pressure drop which is reflected as back pressure on the drain port of the component being drained. Excessive back pressure in draining can result in malfunctioning of valves and damage to seals. Component manufacturer's limitations on maximum allowable back pressure on drain ports should be followed in all cases. If the component being drained is located below the level of the fluid in the reservoir, the pressure due to static head must be added to the pressure drop to determine the total back pressure. In computing the velocity in drain lines, the fluid flow should be based upon the maximum allowable drain flow at which replacement or overhaul is required, rather than the flow from a new component, and should be based upon the viscosity of the fluid at operating temperature.

(o) Pilot lines. Normally the flow through pilot lines is small enough so that velocities will be low. In any event velocities should not exceed 10 to 15 fps.

(p) Hydraulic fluid. A petroleum oil with a high viscosity index should be selected to minimize the change in pipe friction between winter and summer months. The oil selected must have a viscosity range suitable for the system components and their expected operating temperature range. Generally, the maximum viscosity range is between 4,000 Saybolt Universal seconds (SSU) at start-up and 70 SSU at maximum operating temperature. However, this range will vary between manufacturers and types of equipment. In the case of a system containing a large quantity of fluid, rust and oxidation inhibitors should be added. An alternate to the above-recommended hydraulic fluid is a relatively new biodegradable and nontoxic fluid. This fluid uses vegetable-based oils and synthetic additives to provide specific properties which are required in hydraulic fluids. This fluid meets some pump manufacturers' requirements, but it does not meet all manufacturers' requirements, as of this date, so care must be exercised in selecting this fluid. Also, there is only one manufacturer which produces this fluid at present. This fluid, while clear when new, turns amber with age and usage. This may or may not be a problem.